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Lubrication

A Technical Publication Devoted to the Selection and Use of Lubricants

THIS ISSUE

AUTOMOTIVE
HYDRAULIC
TRANSMISSIONS

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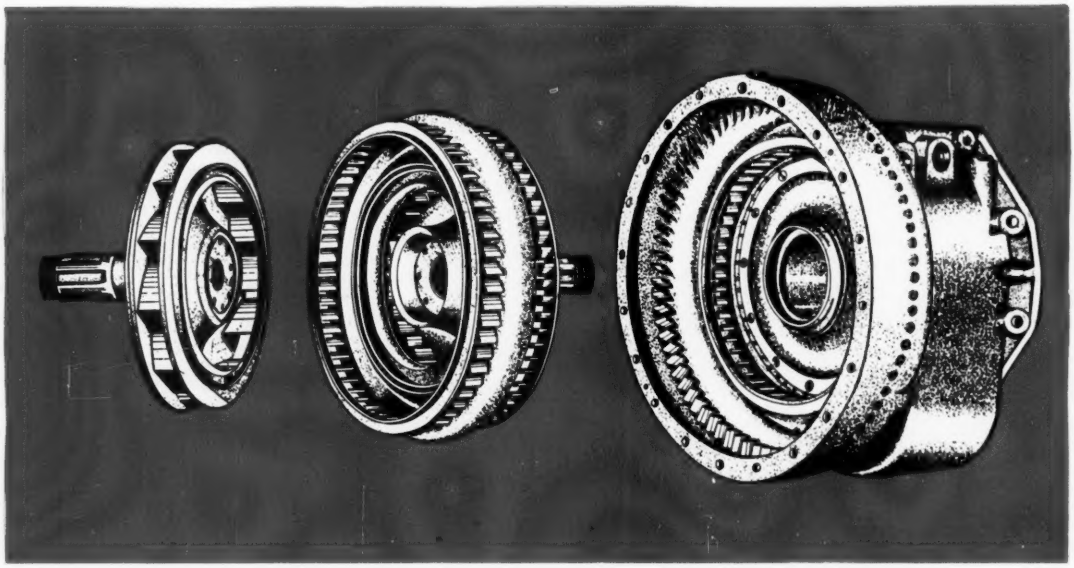

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LUBRICATION

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Automotive Hydraulic Transmissions

The Hydrokinetic Torque Converter

THE ideal motor coach in city operation should be able to accelerate almost as rapidly as other traffic — in other words, as rapidly as the majority of present-day passenger cars — not only to maintain close and profitable operating schedules but to avoid clogging the already congested streets of our larger and older cities.

Such agility in a motor coach is not readily obtainable but it can be approached by decreasing the coach weight, by increasing the output torque of its engine or engines, and by *increasing the facility and efficiency with which engine torque is multiplied and transmitted to the driving wheels.* All of these methods are being used by the motor coach manufacturers: the last is the chosen province and most spectacular demonstration of the abilities of the hydrokinetic torque converter.

The torque converter is not new, having been conceived in 1905 and having served for many years in London's omnibuses. Why then did it not appear sooner in this country? The answer to this question appears to be that the relatively level terrain of London favored the early converter and forgave its early deficiencies. Much additional engineering and development work by American firms was necessary before converters could effectively deal with the difficult terrain, express service and higher schedules so common to American operations.

This development work recognized and included a new concept: that the converter is such an important integral part of the coach that its characteristics must be closely coordinated and fitted to other associated factors such as engine torque, rear axle gear ratios, coach weight and capacity, operating terrain and desired operating schedules. The success of this coordinated engineering is attested by the fact that all of the major American manufacturers of urban motor coaches such as that illustrated in Figure 1, page 118, now provide torque converters. Furthermore, various truck and crawler tractor manufacturers are actively experimenting with them and industrial type converters have been used for several years in switching and freight locomotives, hoisting winches and similar applications where the unique ability of the converter to deliver a steady sustained torque is an outstanding advantage. As inferred by Figure 2, page 119, the torque converter also saw active military service during World War II in the artillery "Tractor, high speed, 38-ton M6" (which used two converters), in its lighter brother, the "Tractor, high speed, 18-ton, M-4", in the "General Pershing" Heavy Tank M6, the "Hellcat" 76mm Motor Gun Carriage M18, and in several other equipments which were completely developed when the war ceased.

In previous issues[1] we have examined the general construction, operation, and illustrative automotive applications of the fluid coupling. We may recall that the fluid coupling consisted of only two sets of blades, both of which rotated, and that its output torque was always the same as the input torque. In other words, the fluid coupling was actually a spe-

[1] "Lubrication" for November 1946, April 1947.

cial clutch which could smoothly transmit but not increase torque.

At the outset, let us admit that the term "torque converter" is technically inaccurate, since nothing can convert torque. More applicable names for our mechanism would be *"power* converter"[2] or "torque *multiplier*". In view of its common usage, however, we will continue to use the term "torque converter".

As illustrated in the "exploded" Figure 3, page 120, the multiple stage hydrokinetic torque converter superficially resembles a more complicated and more highly refined fluid coupling, or better yet, a compact steam turbine. Like the coupling, there is no mechanical connection between the converter's input and output shafts, all torque being transmitted by hydraulic means. Unlike the coupling, however, the most common hydrokinetic converter contains six sets of blades of which two sets (called the "reactors") are fixed solidly to the converter housing and are, therefore, always stationary. The presence of those reactor blades is the major difference between the fluid coupling and torque converter, and is the basic reason for the latter's unique ability to multiply its input torque by as much as 5 to 6 times. Unlike the coupling, but like the steam turbine, the blading of the torque converter is accurately machined all over to a highly developed air foil or stream-line cross section having a high surface finish. The number, shape, dimensions, and angle of these blades all have a pronounced effect on the performance of the converter and are the primary factors used by the transmission engineer in coordinating the converter with the characteristics of the engine to be associated with it.

Courtesy of ACF-Brill Motors Company

Figure 1 — Modern urban motor coach.

As indicated by the term "hydrokinetic", the Lysholm type converter transforms engine power by first converting it to kinetic energy in a stream of fluid, and then reconverting the kinetic energy to mechanical energy at the converter output shaft (2). Since the conversion is done hydraulically, the torque converter has two major advantages over the conventional gear transmission, which is its closest equivalent:

[2]Input × Efficiency = Output HP: Neglecting Dimensional constants, Engine Input Torque × Engine Input Speed × Efficiency = Converter Output Torque × Converter Output Speed. In other words, engine power is applied to the converter as a combination of low torque and high speed and is converted by hydraulic means to much higher torque at much lower speed. If output speed is zero as is the case when the coach is "stalled" with engine running, then the efficiency must also be zero to satisfy the equation and even though the output torque can be quite high.

a. Under "stall" conditions (that is when the motor coach is stationary) and unlike the gear transmission, the engine can deliver its full power to the converter which transforms it into a powerful static output torque that is as much as 5 to 6 times in the engine's input torque.

b. Again unlike the definite shift ratios of the gear transmission, the converter multiplies torque smoothly and wholly automatically without any stops, the conversion proceeding smoothly through an infinite series of torque multiplication ratios between 6 and 0. Simply restated, the operator of a torque converter does no gear shifting.

Figure 4, page 121, represents an oversimplified or linear diagram of a torque converter. Engine power is applied to the blades of a centrifugal pump which accelerates the incoming fluid stream to a high velocity (thereby charging it with kinetic energy) and then directs it against the vanes or follower blading (F-1) of the first output stage. A portion of the kinetic energy is absorbed by the first stage and converted into a thrust having the same direction as the impinging stream. As in any impact, a portion of the kinetic energy is also converted to heat. As discharged from the output or follower blading of the first stage, the fluid still possesses most of its inherent energy but is travelling in a reverse direction, hence the reactor blading (R-1) of the first stage is necessary to straighten it out before it can be properly directed against the output blading of the next stage. Obviously, the stream of fluid playing against the reactor blading also develops a reaction thrust, which is equal but opposite in direction to that impressed upon the first follower blading. The action of the reaction blading has been compared to the fulcrum of a lever: without such a fulcrum, no multiplication of torque can occur.

The combination of a set of follower blading and reaction blading is known as a "stage". Theoretically, a large number of stages would be necessary to completely reconvert the kinetic energy of the fluid stream back to mechanical energy; the practical number of stages, however, has been found to be three. Although apparently omitted, the reaction blading of the third stage is actually the blading of the centrifugal pump.

The overall efficiency of any hydrokinetic fluid system is determined by the characteristics of its design and those of the fluid being pumped: long or undersized channels, the number and severity of any bends or changes in direction, the velocity of fluid and its operating viscosity should all be kept to a minimum, since they either interfere with the efficient application of kinetic energy to the fluid stream, or to subsequent loss of energy from the stream by conversion to heat. The linear system illustrated in Figure 4, page 121 would violate most of these conditions and is therefore impractical. However, by ingenious design, it has been possible to greatly reduce fluid travel and turbulence by the clever arrangement of the blading illustrated in Figure 5, page 121. A comparison between this diagramatic sketch and the photograph of an actual converter in Figure 3 will show that this compact and desirable arrangement is actually accomplished when the pump, turbine-wheel and casing of the converter are assembled. The high order of ingenuity

Courtesy of United States Army

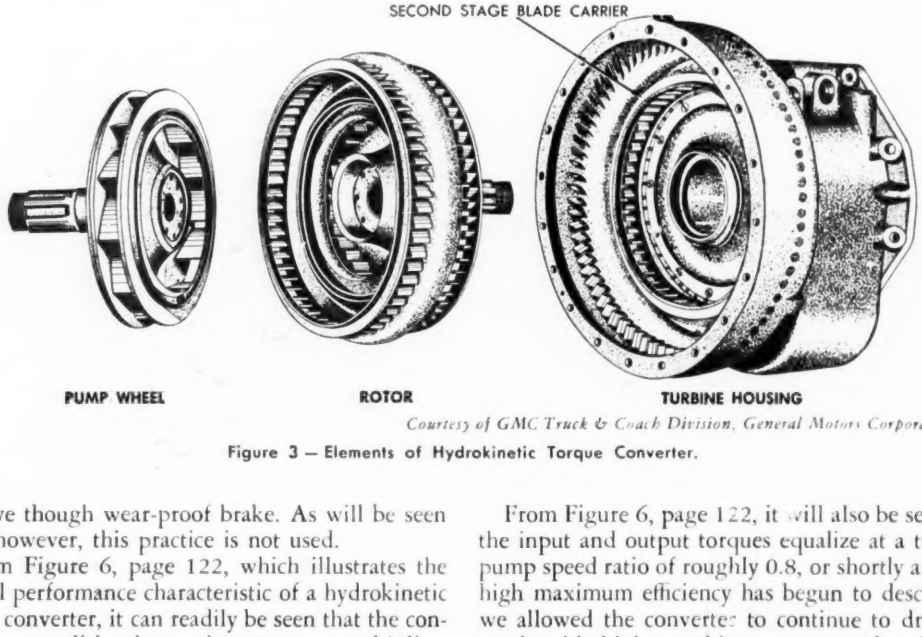
Figure 2 — Tractor, High-Speed, 38-ton, M6.

displayed by the designer in combining six sets of pump, follower and reaction blading in a single compact case makes it admittedly difficult to appreciate their arrangement with a single glance.

As previously pointed out, the converter blading is curved and fixed at a precise angle, which means that the moving elements will rotate in one direction only. Since the same blading angle must serve throughout the considerable and varying speed differences between pump and follower, and follower and reactor speeds, the angle and curvature of the blading must be a nicely adjusted compromise. Since it is impractical to reverse or otherwise change the blading angles, a conventional reverse gear must be provided. If the output shaft were allowed to drive (as by coasting down a hill) the follower blading would attempt to assume the function of the pump which would cause excessive fluid turbulence so that the converter would act as a mildly effective though wear-proof brake. As will be seen later, however, this practice is not used.

From Figure 6, page 122, which illustrates the general performance characteristic of a hydrokinetic torque converter, it can readily be seen that the converter accomplishes its maximum torque multiplication when the motor coach is stationary (and the turbine-pump speed ratio is zero). Under this purely momentary condition, the efficiency of the converter is actually zero for reasons previously discussed (2). "Efficiency" under this condition is a purely fictitious and meaningless quantity since it can never be other than zero. In other words, the converter's prime ability to produce high torque at zero speed is often ignored by the unthinking, who regard the necessary accompaniment of zero efficiency with misplaced horror.

As the converter-equipped motor coach moves and gathers speed the turbine-pump speed ratio also increases and torque applied to the driving wheels automatically decreases with infinite stepless smoothness. This is precisely what the operator of a conventional gear transmission attempts to approximate as he laboriously slips his clutch, accelerates in low gear, double-clutches into the next higher gear, accelerates in that — and so on. This is not efficient. If the torque demanded by the coach should increase (as in approaching a hill) the driver of a conventional gear transmission must go through the same dreary and laborious procedure in the opposite direction. Worse yet, if he is unskillful or careless, he may get caught "between gears" and be unable to get into any, which is a very dangerous predicament for all concerned. On the other hand, the torque converter is always "in gear" and it automatically "changes its gear" to deliver *exactly* the ratio required by the coach at any moment.



Courtesy of GMC Truck & Coach Division, General Motors Corporation

Figure 3 — Elements of Hydrokinetic Torque Converter.

From Figure 6, page 122, it will also be seen that the input and output torques equalize at a turbine-pump speed ratio of roughly 0.8, or shortly after the high maximum efficiency has begun to descend. If we allowed the converter to continue to drive the coach with higher turbine-pump speed ratios, we would suffer real losses. As a consequence, all hydrokinetic torque converters are provided with an auxiliary "mechanical section" whose primary function is to automatically disconnect the converter and at the same time mechanically connect the engine to the driving wheels whenever the input and output torque of the converter are equal. Simultaneously, these mechanisms completely disconnect both the converter pump and followers so as to avoid the unnecessary power losses that would otherwise occur if these elements were permitted to "windmill" uselessly. The mechanical sections also accomplish two other functions:

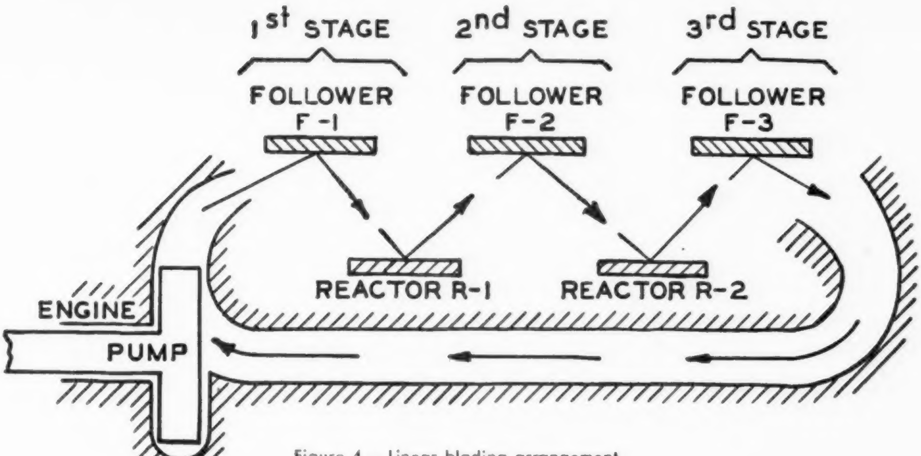


Figure 4 — Linear blading arrangement.

a. They provide a neutral or "disconnect" position of which the converter alone is inherently incapable.

b. They incorporate a reverse gear as previously discussed.

The associated mechanisms and control systems required to accomplish these four functions, and not the basic torque converter itself, are the real complexities of the total mechanism. As an example, and reverting for the moment to Figure 6, page 122, it will be seen that when converter input and output torques are equal, the turbine-pump speed ratio is about 0.8. In other words, engine or pump speed is about twenty per cent higher at the moment of shift than the output shaft speed. As a consequence, all control systems incorporate a device which automatically but momentarily "takes the engine throttle away from the driver" and closes it briefly to equalize input and output speeds so that a smooth shift to direct drive may be easily made.

Since the mechanical sections and control systems used by the several coach manufacturers vary

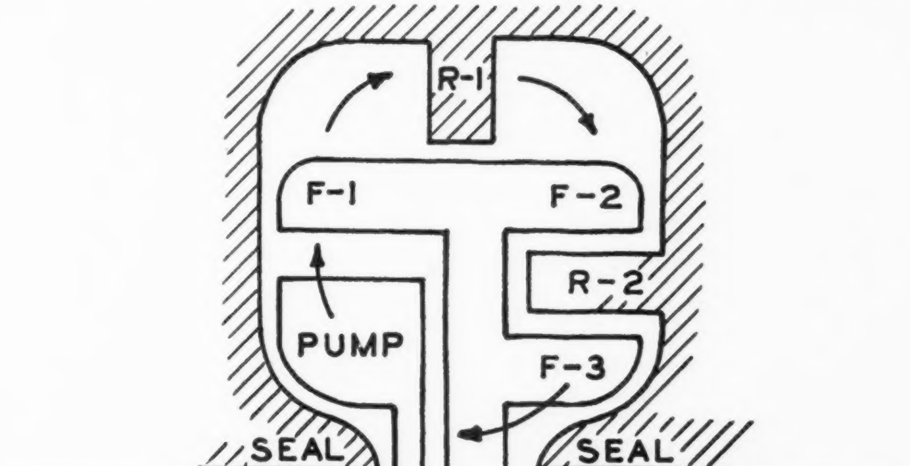


Figure 5 — Curvilinear blading arrangement.

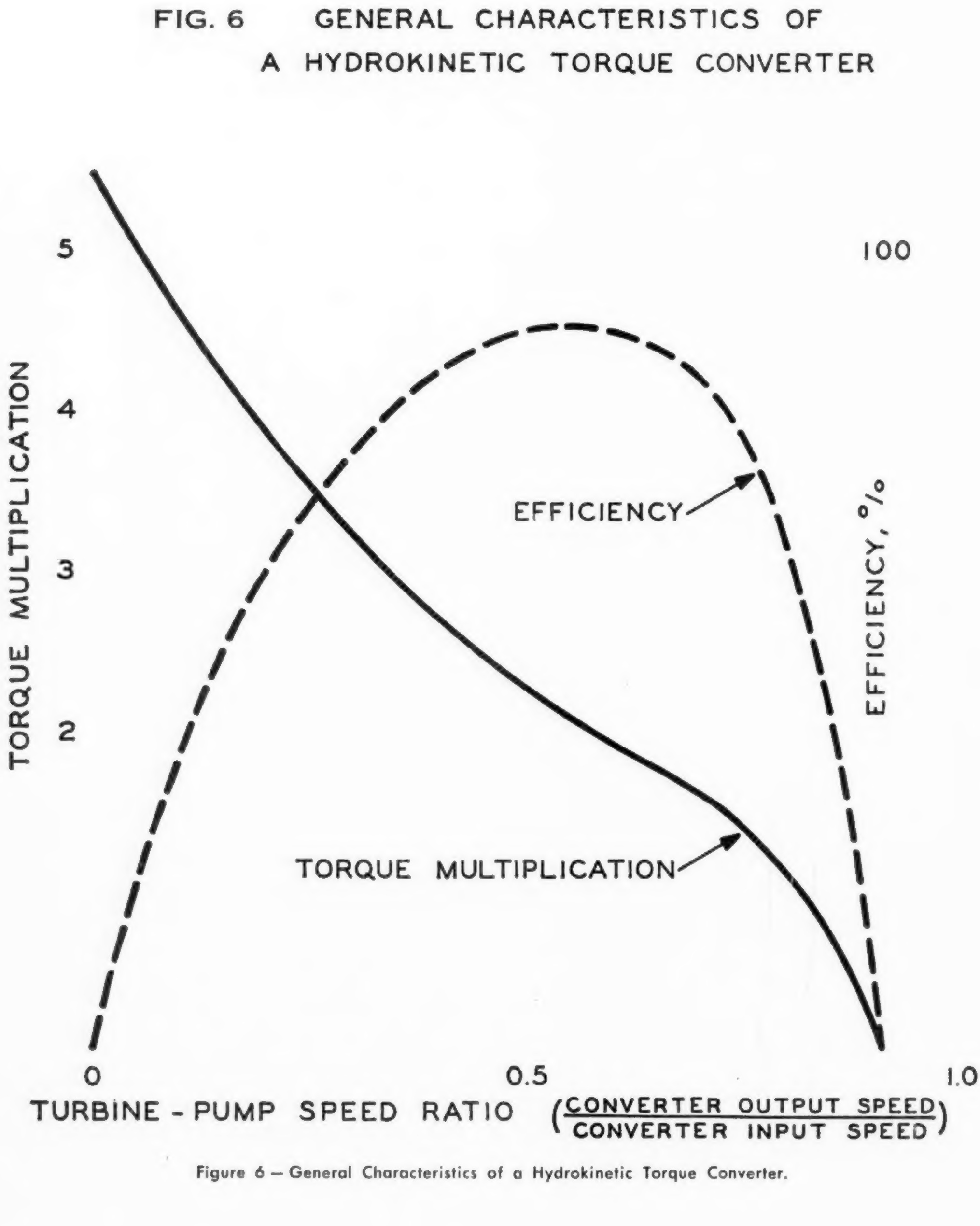
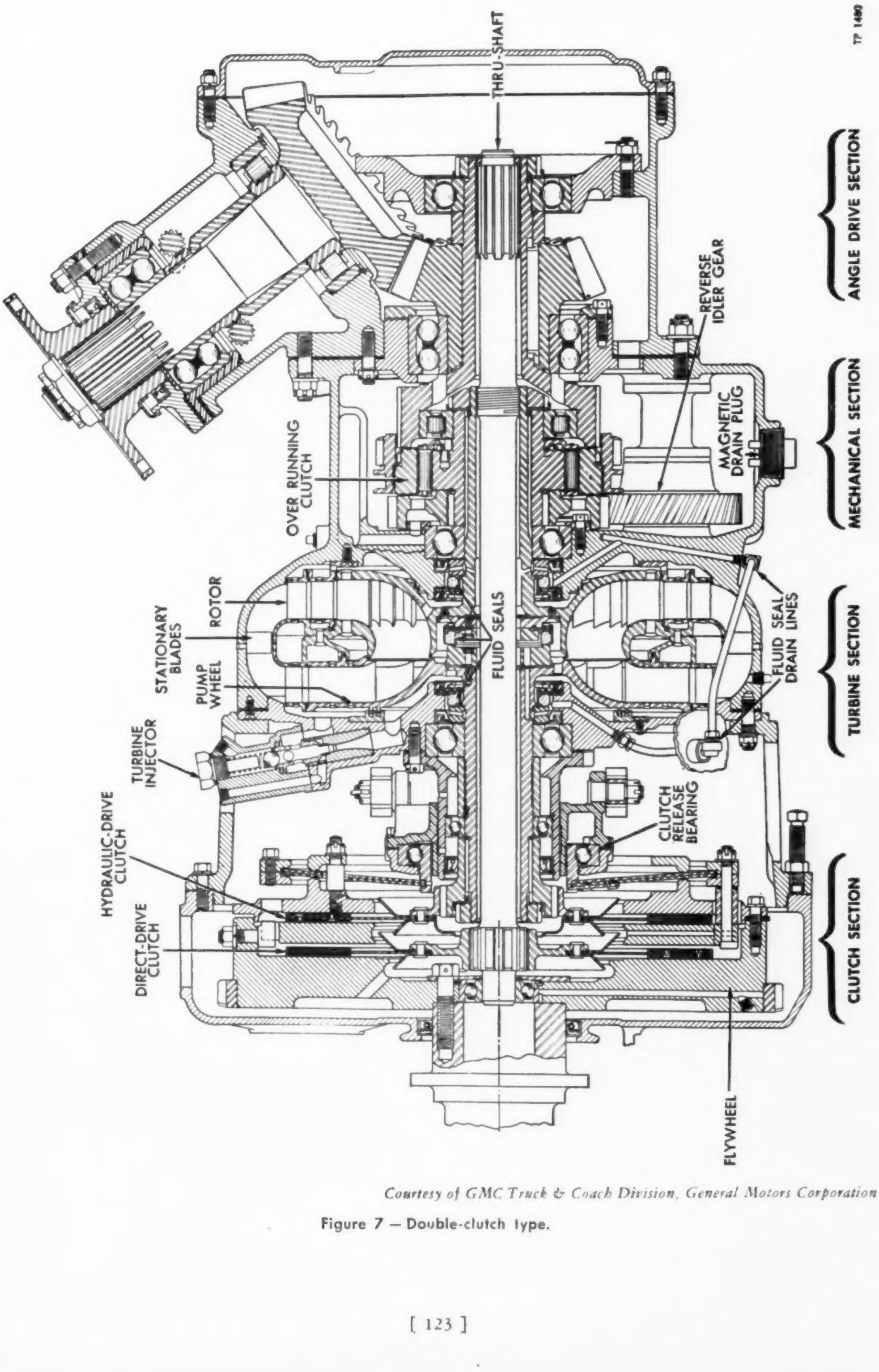


Figure 6 — General Characteristics of a Hydrokinetic Torque Converter.



Courtesy of GMC Truck & Coach Division, General Motors Corporation

Figure 7 — Double-clutch type.

considerably, and since further refinements are constantly being made, it is impractical in this article to attempt more than brief general descriptions[8].

Figure 7, page 123, represents a cross section of a converter having a double clutch type mechanical section, which has enjoyed wide usage. At the outset, let us point out that in this and in the succeeding figure, the "angle drive section" is a convenience of design which is utilized in those motor coaches where the engine is placed across the coach at the extreme rear and either above or immediately adjacent to the driving rear axle. In general, the functions of the converter and mechanical section are not affected by the incidental presence of an angle drive: on the contrary, a large number of the converters now in operation (such as those illustrated in Figure 9, page 127) utilize an output shaft that is coaxial with the engine crankshaft.

Excepting the centrally-located converter, and the angle drive, the most eye-catching detail in Figure 7, page 123 is the ingenious double clutch at the reader's left. This clutch is automatically moved to any of three positions by means of an external air cylinder as demanded by the control system; the friction plates of these clutches cannot both be engaged simultaneously. The unique single conical type clutch spring is used to apply pressure to and engage either clutch plate by "buckling" in the desired direction. In its mid or flat position, the spring disengages *both* clutches and thus provides a transmission neutral.

When the left hand or foremost "direct drive" clutch is engaged, engine torque is transmitted directly and by wholly mechanical means along the through shaft to the angle drive and rear wheels. The over-running or "one way" cam and roller type clutch is used to prevent the follower or turbine rotor from being driven by the output shaft. Since the converter pump is driven by the right hand or rearmost "hydraulic drive clutch", which is disengaged, all hydraulic elements of the converter are stationary and the transmission is in direct drive.

If the direct drive clutch is released, and the hydraulic drive clutch applied (which condition is actually illustrated in Figure 7, page 123) the engine drives the pump of the converter which drives its turbine hydraulically, and the resultant multiplied torque is transmitted through the over-running clutch to the angle drive, and the transmission is in hydraulic drive.

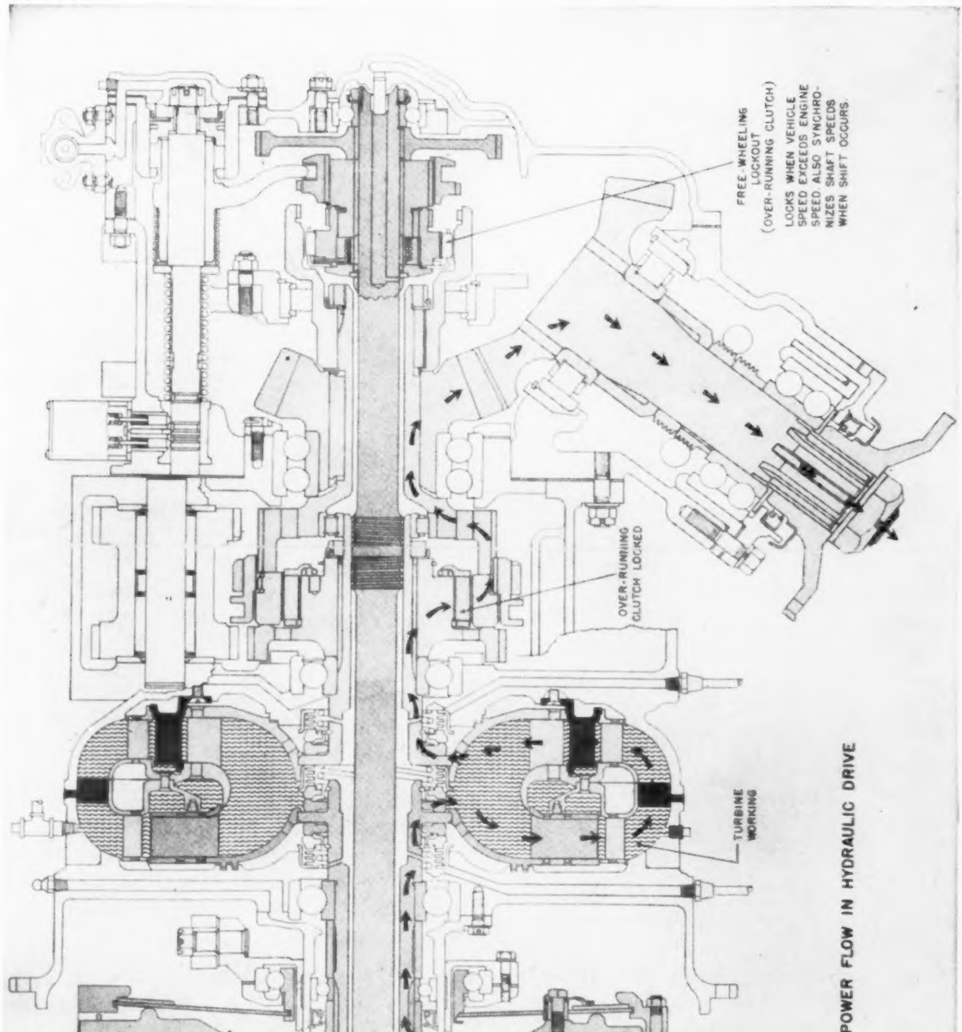
A more minute examination of Figure 7, page 123 will disclose the presence of a toothed or positive dog clutch around the exterior of the over-running clutch which can be moved by the operator. In its forward position (as actually shown in the figure) the dog clutch locks the exterior element of the over-running clutch to the output shaft; in its rearward position, the gear teeth on the exterior of the dog engage the reverse idler gear (not illustrated) and provide reverse drive by hydraulic means.

The electro-pneumatic control system employed with the double clutch converter is normally adjusted to cause an automatic shift from hydraulic to direct drive when the coach speed attains about 21 miles per hour. To prevent "hunting" the control system will not permit the return shift (from direct to hydraulic drive) to occur until the coach has decelerated to about 14 miles per hour. An over-rule switch connected to the accelerator pedal permits the operator to postpone the automatic up shift and to remain in hydraulic drive, which is advantageous when passing another vehicle or ascending a hill at speeds above 21 miles per hour.

Before leaving Figure 7, page 123, the reader's attention is invited to the presence of a "turbine injector" which is provided to insure that the converter is completely filled at all times with the hydraulic fluid at a pressure of 30 or more pounds. For, unlike the oft partially-filled fluid coupling, the converter will not operate at its highest efficiency if either hydraulic cavitation or air entrainment are allowed to occur. The injector utilizes the same operating principle as the familiar steam boiler feed water injector, and obtains its necessary actuating pressure from a connection near the largest diameter of the converter pump. Other models by the same manufacturer now utilize a mechanical engine-driven gear pump for the same purpose. Regardless of the injector or gear pump, the converter is also provided with either an aspirator or a bleed line connected to the highest point of the converter housing which has the sole purpose of removing and exhausting any accumulation of air or fluid vapor.

Figure 8, page 125, illustrates one of the two types of single-clutch converters that are in widespread use. As will be seen from the figure and unlike the converter illustrated in Figure 7, page 123, the mechanical section of this type contains two "one-way", "over-running" or "free wheel" clutches. The first or centrally-located larger one is contained within the slideable forward-reverse selector gear and enables the converter turbine to drive the output shaft without being driven by it. The smaller one-way clutch at the extreme right of Figure 8, page 125 also incorporates a slideable dog clutch with a "balk" mechanism to insure that synchronization of the two elements is attained before the positive dog clutch can be engaged. In effect, this additional one way clutch performs the same function as the additional friction clutch of the double clutch type. It also performs the additional function of providing an all-mechanical connection between the wheels

[8]Those readers who are interested in greater detail are advised to purchase the pertinent literature applicable to the specific model in which they are interested from the coach manufacturer.



Courtesy of Mack Manufacturing Corporation

Figure 8 – Single-clutch type.

and the engine whenever the former drives the latter as during deceleration. This arrangement, of course, enables the engine to exercise its full braking effect during deceleration.

When the vehicle is in hydraulic drive, and as indicated by the arrows in Figure 8, page 125, torque is transmitted through the single friction clutch to the converter pump and thence through the turbine, large over-running clutch, and final drive. It should be noted that the central transmission thrushaft is splined to the engine flywheel, is always connected to the engine, and is driven at engine speed regardless of whether the friction clutch is engaged or not. The small over-running clutch is so designed that the engine torque cannot be transmitted through it to the rear wheels. However, a small reversal in torque (which occurs whenever the coach attempts to drive the engine) causes the spring-actuated rolls to engage their cams and thereby to permit the coach to drive the engine. This action is entirely separate from that of the associated dog clutch described in the following paragraph, and occurs regardless of whether the dog clutch is engaged or not. In other words, this small over-running clutch effectively prevents the coach from free-wheeling at any time and therefore allows the engine to be used as a brake.

At a set vehicle speed, the control system admits compressed air to pistons which —

a. disengage the friction clutch and

b. move the slideable element of the smaller one-way clutch to the left and into positive engagement with its mating female part which is attached to the final angle drive.

The transmission is now in direct drive since engine torque is transmitted by wholly mechanical means along the thrushaft, through the dog clutch, and thence through the final angle drive. Since the pump of the converter cannot be driven by the plate of the friction clutch which is disengaged, and since the large over-running clutch prevents the drive shaft from driving the turbine, all hydraulic parts of the converter are at rest when the transmission is in direct drive.

The third type of converter in popular use is illustrated in Figure 9, page 127, and closely resembles that illustrated in Figure 8, page 125, with two exceptions: a differential reverse gear (K, D, M, E, P, N, Figure 9, page 127) is used, and output torque is taken through an air-engaged dog-type direct drive clutch (G, H, R) placed between the transmission thrushaft F and the output flange L.

Unlike the planetaries previously described (3), the differential type used in this transmission has a "side by side" arrangement in which the external and sun gears (M, N, Figure 9, page 127) are bevel gears of equal diameters with the planetaries (P, Figure 9, page 127) and planet carrier (E, Figure 9, page 127) placed between them. The slideable selector collar D is splined to both the planetary carrier E and the external gear M. When collar D is in the position actually shown in Figure 9, page 127, external gear M and carrier E are locked together and the entire planetary rotates as a unit to provide a solid coupling which transmits turbine torque in a forward direction. The transmission casing is provided with an additional set of splined teeth "K". When collar D is manually shifted still further to the reader's left, it disengages the teeth on external gear M while engaging the teeth K on the transmission case with the result that the planetary carrier E is held stationary and sun gear N is forced to rotate at the same speed but reverse direction from the driving external gear M.

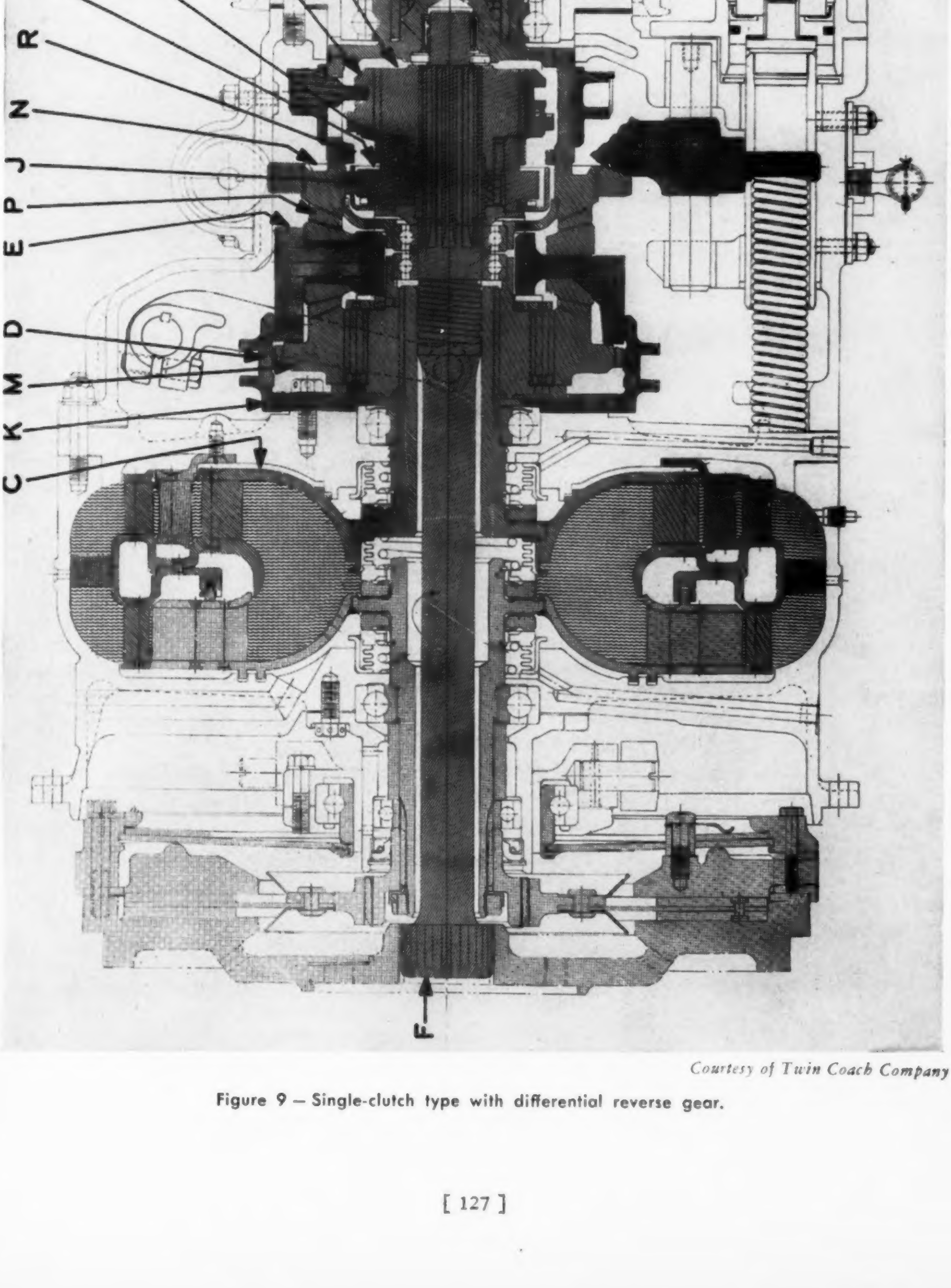
A large one-way clutch inside external gear M permits turbine C to drive the external gear, but prevents transmission or torque in the opposite direction. A smaller one-way clutch J is interposed between thrushaft F and output shaft L which permits the coach to drive the engine during deceleration when coach speed has fallen enough (18 miles per hour) to cause an automatic shift into hydraulic drive. During acceleration and when the coach has attained the usual shift speed of 20 miles per hour, the control system actuates an air cylinder T which moves jaw clutch ring H to the reader's left and into engagement with internal teeth R on jaw clutch housing N. Since the female element G of the jaw clutch is splined to the thrushaft F, the latter is now mechanically and directly connected to output shaft L and the transmission is in direct drive. The internal element of one-way clutch J is separate from G and is allowed a limited radial movement with respect to it, but is kept to one side of its movement range by means of a spring. A series of teeth Q on this element prevent the jaw clutch ring H from engaging with the internal teeth H of the housing until a torque reversal takes place, all teeth are in alignment, and synchronous speeds have been attained.

LUBRICATION

The hydrokinetic torque converters described in the foregoing are all divisible into the following four groups, each of which requires separate and different lubricants:

1. The pilot and throwout bearings of the friction clutch or clutches; the front and outer ball bearings of the converter pump.
2. The converter, converter seals and charging pump.
3. The mechanical transmission.
4. Air cylinders, linkages, and associated elements of the control system.

The first group comprises three grease-lubricated deep-groove ball bearings, all of which rotate at



Courtesy of Twin Coach Company

Figure 9 — Single-clutch type with differential reverse gear.

engine speed under both thrust and radial loads and are continually exposed to normal temperatures of about 250°F. which under extreme stress can rise as high as 300°F. Experience has shown that the maximum life of these bearings is obtained through the regular application of moderate amounts of a high melting point smooth-textured sodium base grease having high resistance to leakage and complete freedom from non-lubricating and abrasive fillers. While National Lubricating Grease Institute Grade 2 grease may be used under moderate operating conditions, the NLGI Grade 3 grease is better able to withstand heavy duty and is therefore preferred. Some greases meeting U. S. Army Specifications 2-110 (Amendment 4) and 2-108B are representative of the type required.

The lubrication of the second group, that is the converter with its seals and charging pump, was originally attempted with kerosine, No. 2 household fuel oil, or diesel fuel. In diesel-engined coaches, the converters were formerly connected to the engine fuel tank. Further experience gradually demonstrated that these non-specialized materials left much to be desired, and in several instances caused maintenance and operating difficulties that were unjustly blamed upon the then "new-fangled" converter. Being non-specialized, or intended for other uses, these materials either entirely lacked or had insufficient capacity in those qualities which are now recognized as essential and perhaps peculiar to the torque converter. For example, and because high fluid temperatures can occur, most of these materials oxidized excessively if used no more than 10,000 miles, and deposited sludge or varnish on the exquisitely machined blading of the converters, or blocked critical passages in radiators, heat exchangers, injectors, filters and relief valves. Worse yet, and since none of the materials possessed appreciable solubility, the deposits could build up and finally require the complete disassembly of the converter for an expensive manual cleaning. Most of these non-specialized materials were of insufficient viscosity or "lubricity"[4] to provide more than marginal lubrication of pump rotors, pump glands, and converter seals with the result that excessive wear and leakage could occur and expensive replacements necessitated. Some of these materials contributed to cavitation and poor torque transmission, and even created a fire hazard by either vaporizing in the turbine system or by entraining air and foaming. Occasionally these materials would etch or chemically attack the various metals in the turbine system. None of them were capable of preventing rust formation from extraneous atmospheric water that condensed in the converter systems of irregularly used coaches.

[4]An early partial solution to the lubrication problem was attempted by adding five per cent volume of lubricating oil to the kerosine or diesel fuel, but this was later abandoned in favor of straight kerosine.

From the foregoing it is now easy to recognize the necessity for specially designed "torque fluids" that could overcome the several serious difficulties. Since even a high quality straight mineral oil is incapable by itself of fulfilling all requirements, the most successful torque fluids are blended from such an oil and two or more additives. Several such fluids are now commercially available: one in particular which has been subjected to extreme tests and approved by the converter and all major coach manufacturers has the following general characteristics by laboratory tests:

Gravity	26.4
Flash, PM °F.	300
Viscosity, SUS at 100°F.	63.1
Viscosity, SUS at 130°F.	47.5
Viscosity, SUS at 210°F.	35.1
Pour ASTM, °F.	—45
Foam Test (CRC L-12-445)	
(Millilitres of foam remaining after ten minutes)	
Sequence 1	0
Sequence 2	0
Sequence 3	0
Rusting Test	
Distilled Water	No rusting
Salt Water	No rusting

It seems unwise at this time to assign more than casual interest to laboratory tests such as the above and to laboratory type specifications that are based thereon. Performance in the torque converter itself should be the sole criterion, particularly since the correlation (if any) between laboratory tests and converter performance is unknown at this time. For example, good oxidation stability in the converter is a prime requisite, but we are not yet certain that any of the usual laboratory oxidation tests that are currently used will surely measure and predict this quality in a torque converter. Field performance is the best criterion at any time, and the only one available to us at this moment.

The various elements in the third group or mechanical section of the converter were designed to utilize a good quality engine oil having a viscosity of approximately 95 SUS at 210°F. Continuing experience indicated that the high viscosity index and high oxidation stability of airplane engine oils were advantageous in some instances. At the present time, all coach manufacturers either directly recommend or permit the use of the SAE-50 grade of a heavy duty motor oil which meets U. S. Army specification 2-104B. Ordinary *gear* oils and hypoid gear oils are not to be used.

Air cylinders of the fourth group are variously lubricated with non-petroleum hydraulic brake fluid, graphite grease, or engine oil (3) depending upon the specific design and manufacturer.

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